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ASSESSMENT OF HEAT PUMPS FOR DISTRICT HEATING APPLICATIONS

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ABSTRACT

Heat pumps will be a major player in the future energy system for their ability to efficiently extract heat from a source at lower temperature and provide it at higher temperature using electrical work. If coupled with a heat storage tank, the system can store heat and provide it at the point in time when the electricity price is more favorable.

In this study, we present a model of the heat pump energy performance using the coefficient of performance (COP) and the Lorentz efficiency. The latter gives an indication of the offset of the actual COP from the theoretical COP. We then compare different refrigerants performance to find out which one provide the best performance for district heating application. Finally, we study the cost-optimal operation of a heat pump with and without a thermal storage tank.

Results show that ammonia is a better candidate in terms of performance compared to other selected refrigerants. The performance analysis shows that the heat pump has a better Lorentz efficiency for lower COPs. Finally, we show that coupling the heat pump to a thermal storage tank can reduce the electricity cost of operation.

Keywords: Heat pumps; COP Lorentz Efficiency.

NONMENCLATURE

Abbreviations	
DH	District heating
HP	Heat pump
T_{LM}	Logarithmic mean
VHC	Volumetric Heating Capacity

1. INTRODUCTION

Different options have been considered to provide heat and electricity for domestic use in the future energy system. This may include fuel cells [1], solar collectors [2], and heat pumps. Heat pumps can provide heat at the desiderate temperature without producing CO_2 if driven by electricity from renewable sources.

Heat pumps work according to the reverse Rankine cycle. The main components in the cycle are a compressor, an expansion valve, an evaporator and a condenser. However, a combination of multiple cycles is seen in some cases. In the internal circuit, a refrigerant goes through repetitive compressions and expansions. During the compression phase, heat is released to the sink while during the expansion heat is absorbed from the heat source [3].

With the aim to select a feasible refrigerant for district heating applications in this paper we develop a model of a single-cycle heat pump with different selected refrigerants. We then model the heat pump coupled with a heat storage tank in EnergyPRO. The software calculate the cost-optimal operation of the system with electricity price traded on the Day-Ahead spot market.

2. MATERIAL AND METHOD

The heat pump model was developed in Python using the libraries Coolprop [4] and Numpy. The cost optimization of the heat pump with and without the heat storage tank was developed in the software platform EnergyPRO [5]. In the following sections, the heat pump model is presented.

2.1 The Lorentz COP

The heat pump, theoretical energy performance is usually calculated using the Lorentz COP as described in the equation below:

$$COP_{Lorenz}=\frac{T_{LM,H}}{T_{LM,H}-T_{LM,C}}=\frac{T_{LM,H}}{T_{lift}} \quad (1)$$

This COP value is based on a Lorentz cycle in which the heat absorption/rejection occurs by a nonisothermal process with same temperature change for both the heat source and sink. In the cycle compression and expansion are isentropic therefore the all cycle process is considered reversible [1]. The COP_{Lorenz} only depends on the source and sink temperatures and not on the characteristics of the heat pump (eg refrigerant type, evaporator and condenser, expansion valve, number of cycles, pressure drops). In the equation $T_{LM,H}$ and $T_{LM,C}$ are defined as the logarithmic mean temperature of the hot and cold stream:

$$T_{LM,H}=\frac{\Delta T_H}{\ln\frac{T_{H,o}}{T_{H,i}}} \qquad T_{LM,C}=\frac{\Delta T_C}{\ln\frac{T_{C,i}}{T_{C,o}}} \tag{2}$$

with $T_{LM,C}<T_{LM,H}$.

The temperature lift is the difference between the logarithmic mean temperatures in equation (2):

$$T_{Lift} = T_{LM,H}-T_{LM,C} \tag{3}$$

The COP increases when the temperature lift, T_{Lift}, is reduced i.e. when the source and sink temperature are closer with each other. In other words, the heat pump performs better with small temperature lifts.

In figure 1, the temperature levels of the refrigerant and source/supply are shown on the TS diagram. In practice, a temperature glide of the refrigerant is seen during both condensation and evaporation, which causes a temperature variation during these two stages.

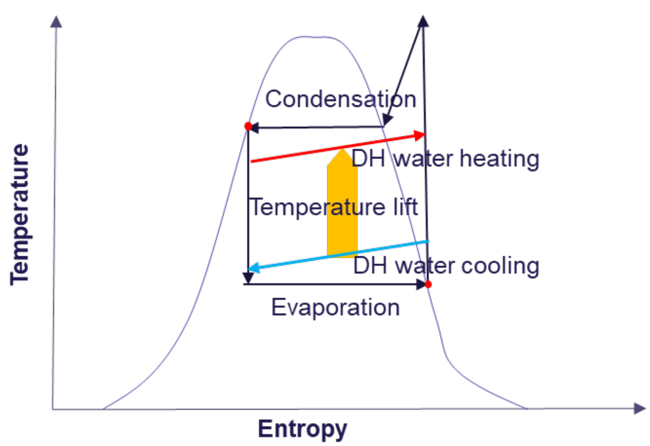

Figure 1. Refrigerant TS diagram with temperature levels of the heat source and supply. The 2 red dots are the positions where it is assumed the end of condensation and evaporation processes

Figure 2 shows the Lorentz COP values for different Log-mean temperatures of source and sink. A sharp increase of the COP values is seen when the source mean temperature increases while the sink mean temperature reduces hence reducing the temperature lift.

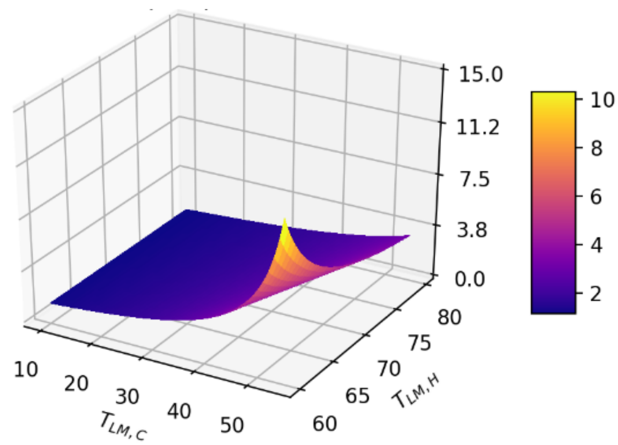

Figure 2. Lorentz COP at different $T_{LM,H}$ and $T_{LM,C}$

The heat pump cycle is limited by system process irreversibilities (e.g. compressor and other components efficiency, heat losses) and refrigerant limitations (e.g. critical temperature and pressure). Therefore a factor called Lorenz efficiency, $\eta_{Lorentz}$, is introduced and it is defined as the ratio of the actual heat pump COP, usually provided by the manufacturer, compared with the theoretical Lorenz COP, that is, the ratio between the useful output and the reversible heat output.

$$\eta_{Lorentz} = \frac{COP_{HP}}{COP_{Lorentz}} \tag{4}$$

In this equation, the COP_{HP} is the ratio of the heat pump heat capacity and the compressor consumption.

$$COP_{HP} = \frac{\dot{Q}_{HP}}{\dot{W}_{Comp}} \tag{5}$$

The heat provided by the HP is the sum of the heat of condensation and the compressor work.

$$\dot{Q}_{HP} = \dot{Q}_{Cond} + \dot{W}_{Comp} \tag{6}$$

The compressor work is defined as the product of the isentropic compressor work and the compressor efficiency.

$$\dot{W}_{Comp} = \eta_{Comp}\dot{W}_{isen} \tag{7}$$

The compressor efficiency, η_{Comp}, was fixed to 0.5 and the isentropic compressor, $\dot{W}_{isen}$, assumes the compressor process isentropic.

In the model calculation, it is assumed that the refrigerant reaches zero "vapor quality" at the end of the condensation and "vapor quality" equal to one at the end of the evaporation. This is shown in figure 1 with two red dots.

Refrigerants with larger heating capacity require less compressor power as less mass flow is needed to produce the same amount of heating. To analyze this refrigerant property we introduce a parameter called "volumetric heating capacity" which is the product of the density of the refrigerant at the compressor inlet and the heat realized by the heat pump. We define this value per unit of volume [6].

$$VHC = \dot{Q}_{HP}\rho_{ref} \quad (8)$$

2.2 Comparison between feasible refrigerant for HP

The study considers different refrigerants mainly used in industrial applications and for water heating. They are listed below with their main characteristics.

R410a It have been used in several application from water-cooled to heat-cooled heat pump.

R1234ze(E) is a medium pressure refrigerant which has been used in commercial building and supermarkets. It is considered a substitute for R134a [7].

R290(Propane) is used in heat pumps for water heating as a natural refrigerant to substitute R134a.

R717 (Ammonia) is a common natural working fluid for heat pump. It is characterized by high latent heat which is evident by the magnitude of its saturation vapour curve compared to other refrigerants. This is reflected in smaller refrigerant flow rates. The vertical saturation curve during the gas compression shown on the P-h diagram shows that there is limited enthalpy change during the compression and therefore less work.

Table 1. Properties of some common refrigerant for heat pumps

Fluid	Boiling point (°C) at Patm	Critical temperature (°C)	Critical pressure (bar)
R717	-33.3	132.2	113.5
R290	-42.1	96.7	42.5
R410A	-48.5	70.1	49.3
R1234ze(e)	-18.95	109.4	101.1

3. RESULTS

3.1 HP modelling results with selected refrigerants

In this section, the main results produced by the model are plotted. First, we analyzed the saturation curves on the TS and PH diagrams; afterward we analyzed the heat pump performance operations with the following assumptions:

- The evaporator temperature ranges between 10°C and 20°C and the condenser outlet temperature of 70°C.
- Pinch temperatures of 5°C at the evaporator and the condenser. These temperature values should be representative of those found in district heating.

In figure 2 we can see, the comparison between of the saturation curves of the identified refrigerants. All the selected refrigerants can operate the cycle below the critical temperature values. All refrigerants, except for R1234ze(E), can be considered as "dry" as the isentropic compression process from start to end is in the vapor phase. R1234ze(E) shows an almost vertical saturated vapor branch of the saturation curve and therefore is classified as a "isentropic" refrigerant. This implies that they need to be superheated at the evaporator outlet in order to avoid liquid slugging in the compressor.

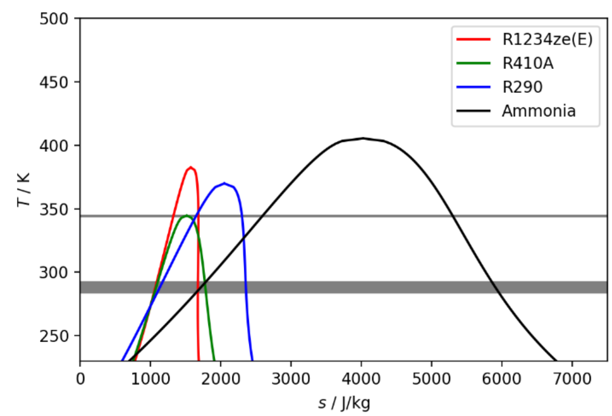

Figure 4. Saturation curve for selected refrigerants on the TS curve

The plots of the saturation curves on the PH diagram in figure 5 are important because from the size of the saturation dome indicates the amount of heat (ΔH) which is released during the refrigerant condensation. From the figure, ammonia has the largest heat release during the phase change. Also, the working pressure of the refrigerants can be inferred.

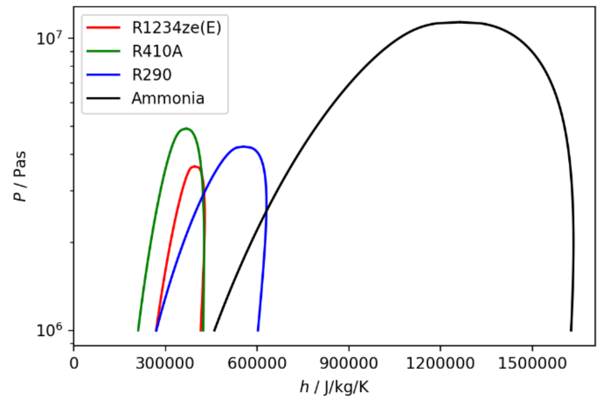

Figure 5. Refrigerants saturation curves on the PH diagram

Figure 6 shows the volumetric heating capacity of heat pump cycle for the mentioned working fluids. The VHC depends on the density of the working fluid and the heating effect. It is useful to indicate the volume of refrigerant necessary to produce the heating effect. From the figure is evident that the VHC values of ammonia and R410A are higher compared meaning that they produce less heat for unit volume.

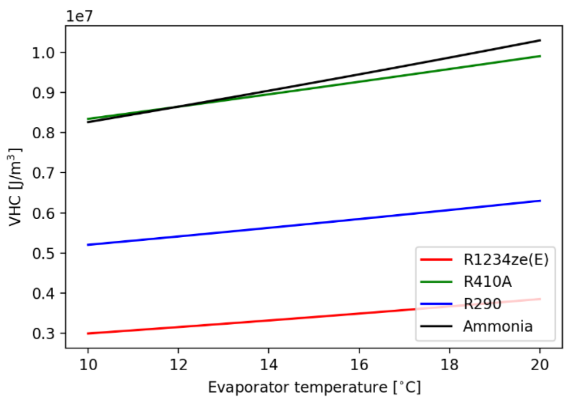

Figure 6. Refrigerants volumetric Heating capacity (Tcond=70°C)

The ΔP during the compression it is useful to estimate the compressor work. Different cycles must to operate at different pressure difference for the same source and sink temperatures. From figure 7, it appears a large variation of ΔP across different refrigerants.

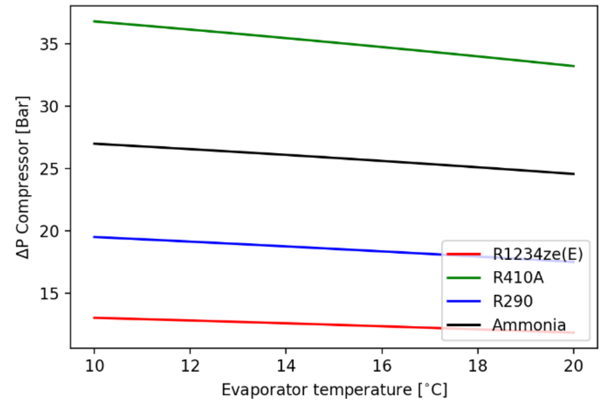

Figure 7. Compressor pressure difference (Tcond=70°C)

Finally, the compressor work is function of the enthalpy difference and the compressor efficiency at the specified cycle temperatures interval and will contribute the heat pump heat generation. Plots of the cycle $\dot{W}_{Comp}$ are shown in figure 8.

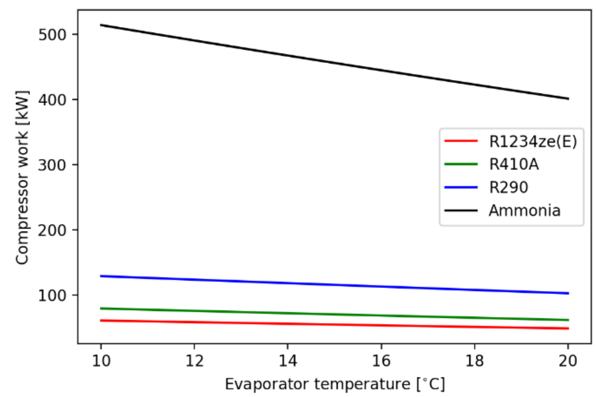

Figure 8. Compressor work (Tcond=70°C)

The heat pump modelling leads to the COP values shown in figure 9. It can be seen that Ammonia has the largest values of COP among the selected refrigerant. In addition, it is evident that the COP increases when the temperature lift is reduced. The figure offers also a comparison of the real COP with the Lorentz COP, which is the one that produces the largest values. This is attributable to the fact that the Lorentz COP assumes isentropic processes and does not include the specific characteristics of the refrigerants.

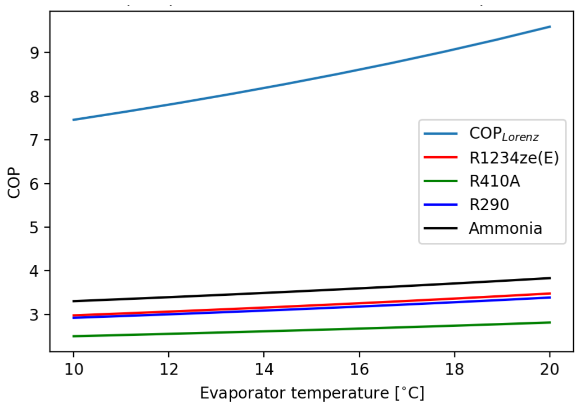

Figure 9. Heat pump COP (Tcond=70°C)

Figure 10 show the Lorentz efficiency calculated as $\eta_{Lorentz} = COP_{HP}/COP_{Lorentz}$. It is evident that the Lorentz efficiency is higher in the case of ammonia refrigerant. The reason is attributed to the fact that a larger fraction of heat is produced in the compression process compared to other refrigerant as shown in figure 8.

Besides, it appears that the $\eta_{Lorentz}$ decreases with the increase of the temperature lift. In other words, real COP values are closer to the ideal COP values for larger temperature lifts. This can be explained with the fact that the compression work is larger for higher temperature lifts and therefore the compression accounts for a larger portion for the heat generation compared to the condensation process.

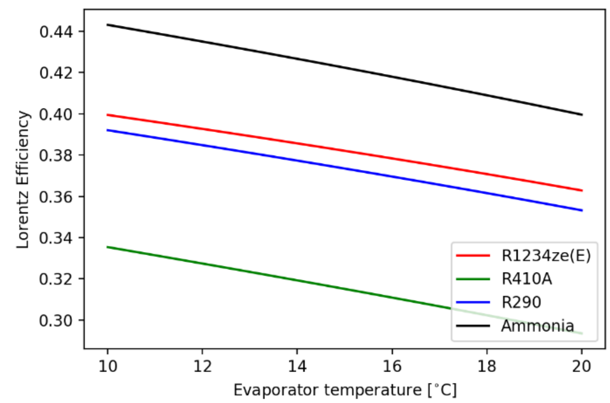

Figure 10. Lorentz efficiency, $\eta_{Lorentz}$ of selected refligerants (Tcond=70°C)

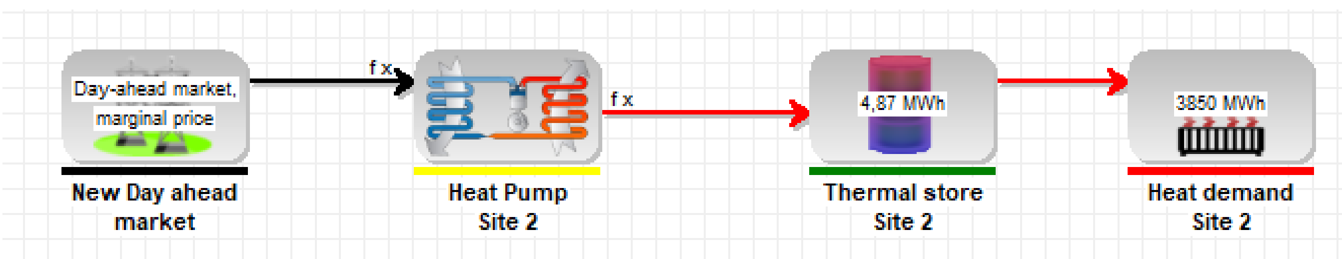

Figure 11. Example in EnergyPRO software platform of a heat pump coupled with a thermal storage tank for distric heating with electricity price traded on the Day-Ahead spot market

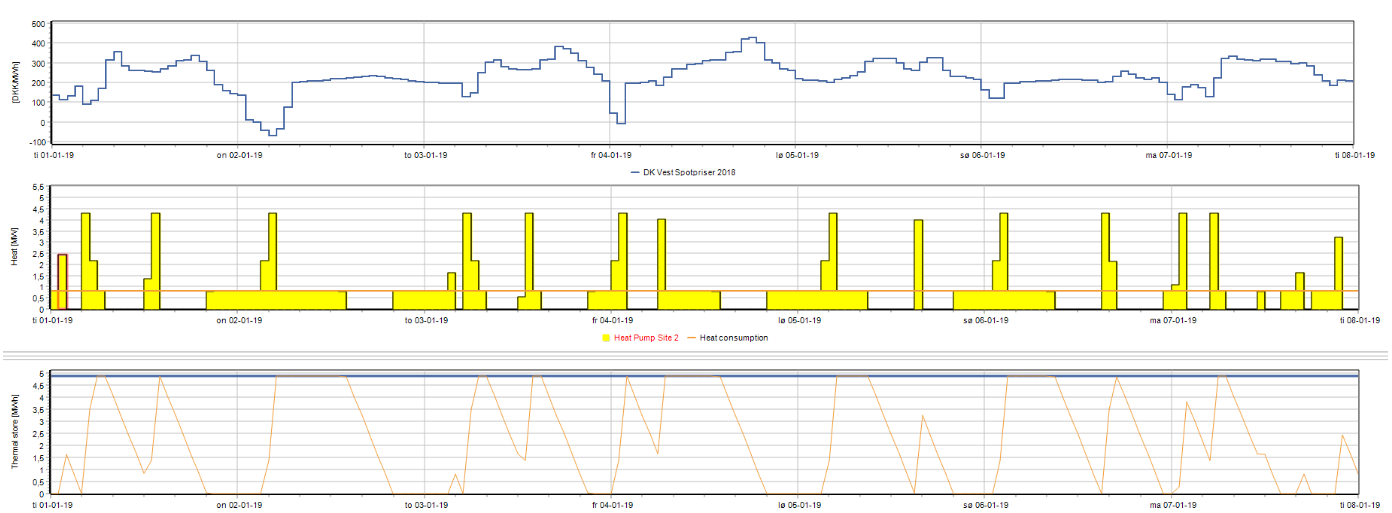

Figure 12. One week operation in EnergyPRO software platform of a heat pump with a thermal storage tank. *Top*. electricity price fluctuations. *Middle*. Heat produced by the heat pump. *Bottom*. Thermal storage level in the tank.

3.2 Cost optimization and comparison HP with and without thermal storage tank

The heat pump model was coupled with a thermal storage tank in EnergyPRO to simulate the cost savings in a situation where there is a continuous change in the electricity price due to short term trades in the electricity market. In figure 11, we can see the model diagram as shown in EnergyPRO. The configuration with the thermal storage was compared to a configuration of a heat pump operating without the thermal storage. In figure 12, we show the results with the cost optimization based on the electricity cost in Denmark during one week. The heat storage tank gives the possibility to use the heat pump when the electricity cost is lower.

We can assume that the cost of operation is affected by the electricity cost and the size of the thermal storage and the heat pump. In the case study, the electricity savings for using the thermal storage was in the range of 61% percent. Besides the number of hours of operation was reduced of 70% on the yearly based compared to a case without the thermal storage.

4. DISCUSSION

While the heat pumps have higher COP for lower temperature lifts, their Lorentz efficiency can show a different behavior. In this study, we compared the actual COP with the Lorentz COP, which represents the ideal maximum COP that can achieved by a heat pump. The actual operating cycle is affected by irreversibilities that prevent the system to reach the max ideal performance. Similar studies were conducted before led to same conclusions. For instance in [8] energy and exergy losses were calculated on each component of the heat pump.

The heat storage tank is crucial to take advantage of a favorable electricity price in situations of intraday fluctuations of the electricity cost leading to significant electricity cost reductions over the year.

5. CONCLUSIONS

In this paper, we have developed a simple model for comparing performance of refrigeration single cycle. While in the majority of the cases heat pump performance are identified using the COP values, the Lorentz efficiency indicates in which point of operation the heat pump provides the best energy efficiency. Consequently, we have proven and quantified the benefits of using thermal storage tank in the contest of variable electricity price.

While this model produces reasonable results for the future, the model can be expanded by including more detailed modelling description of the heat pump components.

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